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Structural Analysis of the Cannon-Caliber Electromagnetic Gun (CCEMG) Integrated Launch Package (ILP)

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Christopher J. Jaeger

ARL-TR-482 July 1994

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13. ABSTRACT *(Maximum 200 words)*

At the direction of the program manager of the Cannon-Caliber Electromagnetic Gun (CCEMG) Program, the U.S. Army Research Laboratory (ARL) was tasked to provide an independent assessment of the work being performed by Kaman Sciences Corp., under subcontract to United Defense (formerly FMC), on this program. The ARL independent analysis focused on aerodynamic, aerothermal, structural and current, and heat transport issues associated with the integrated launch package (ILP) design for CCEMG. This report examines the structural viability of an electromagnetically launched projectile from the CCEMG, using three-dimensional finite element (FE) techniques. The report summarizes what was presented to the CCEMG contractors at the ARL CCEMG ILP Review briefing in July 1993.

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1. INTRODUCTION

Finite element (FE) techniques have been extensively employed in the investigation of in-bore projectile structural integrity (Rabern and Bannister 1991; Kaste and Wilkerson 1992; Burton 1993). For conventional propulsion gun systems—that is, those having the projectile accelerated by gas pressure—a generally accepted technique is to perform a quasi-static analysis (Sorensen 1991). Such an analysis employs a force balance between the gas pressure and the inertial acceleration setback load. Standard practice dictates that the peak pressure load be utilized when performing the analysis to subject the launch package to a worst-case loading.

For the case of electromagnetically launched, solid armature driven projectiles, there is no gas propulsive force. Typically, system specifications are written such that a peak acceleration capability is defined. Previous analysis of a cannon-caliber projectile had used pressure instead of the electromagnetic (EM) body force load to attain an equivalent maximum acceleration (Burton 1993). In that case, this technique was felt to be an accurate representation of the loading since the projectile armature was aft of the sabot body such that the EM force loading was concentrated at the rear. Such an arrangement lends itself to the analogy of a base-pushed conventional design analysis.

The projectile design developed jointly by Kaman Sciences Corp. and the Center for Electromechanics-University of Texas at Austin (CEM-UT) for the Cannon-Caliber Electromagnetic Gun (CCEMG) Program has a mid-riding armature/sabot configuration. CEM-UT was responsible for the initial structural and electromagnetic design and sizing of the discarding armature. Kaman Sciences had responsibility for the detailed structural analysis of the Integrated Launch Package (ILP) and the design and sizing of the subprojectile. Use of a gas pressure equivalent to model the projectile EM load would not accurately reflect the force distribution throughout the projectile body. Therefore, it was decided to adopt a body force loading technique. This was easily accomplished with the use of the ANSYS (DeSalvo and Gorman 1989) FE analysis code which allows for nodal force loading. The gas pressure equivalent method was also done to show the difference in the results for the two load representations.

The CCEMG ILP analyzed in this report is the first iteration of an on-going developing design (Zielinski 1993). Consequently, not all of the dimensional details are current nor applicable to those in the final delivered machine shop drawings.

2. ILP GEOMETRY

The CCEMG ILP consists of a tungsten penetrator core with a titanium fluted-flare attached for aerodynamic stabilization (Figure 1). The subprojectile is supported in-bore by the armature/sabot (Figure 2). The armature configuration utilizes the "tandem contact" concept which has two separate armature surfaces in contact with the rail. The sabot's design has some rather unique features which increased the complexity of developing the FE geometry model.

For instance, the forward borerider has a v-shaped cut recessed into its front face which slopes to the rear and produces some sharp angles in the model. These angles made it difficult to model the sabot using only eight-noded brick elements while preserving a good aspect ratio within the elements. This was especially true along the line where the V-cut intersected the cylindrical penetrator. The line along this resulting intersection was a helical ellipse, and only with great care and effort was it possible to have elements along the intersection that did not exhibit excessive twisting or have perverse aspect ratios. Likewise, the aft ramp of the sabot, which has a hexagonal cross section, has very narrow elements which were carefully tailored so as not to violate aspect ratio requirements.

To simplify the modeling, advantage was taken of projectile symmetry, so that only a quarter of the ILP was modeled with two views of the FE geometry shown in Figures 3 and 4. The conical nosetip has been converted to a cylinder with equivalent mass and appears orange in the figures. Similarly, the titanium fluted-flare is incorporated as a lumped mass equivalent inside the hollow flare hub (as is shown by the green elements in Figure 4). The fluted-flare is positioned in such a way so that the model's center of gravity coincides with that of the actual subprojectile. Use of the lumped mass equivalents for nosetips and rear stabilizers is a common practice that simplifies the geometry model while still accurately reflecting the stresses developed through the rod (Rabern and Bannister 1991).

The grooved interface between the penetrator and sabot was modeled using the average density between the two materials, tungsten and aluminum. The elastic modulus of the more compliant material, in this case aluminum, was selected for the interface elements. The sabot and rod were rigidly attached since no sliding was allowed between the two with the material properties taken to be those at room temperature. Table 1 provides a breakdown of the mass by components as calculated by the ANSYS code.

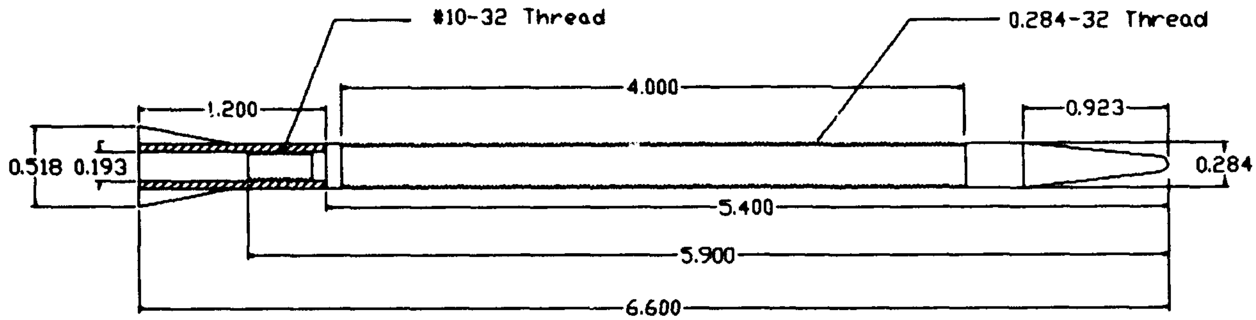

Figure 1. CCEMG penetrator and fluted-flare.

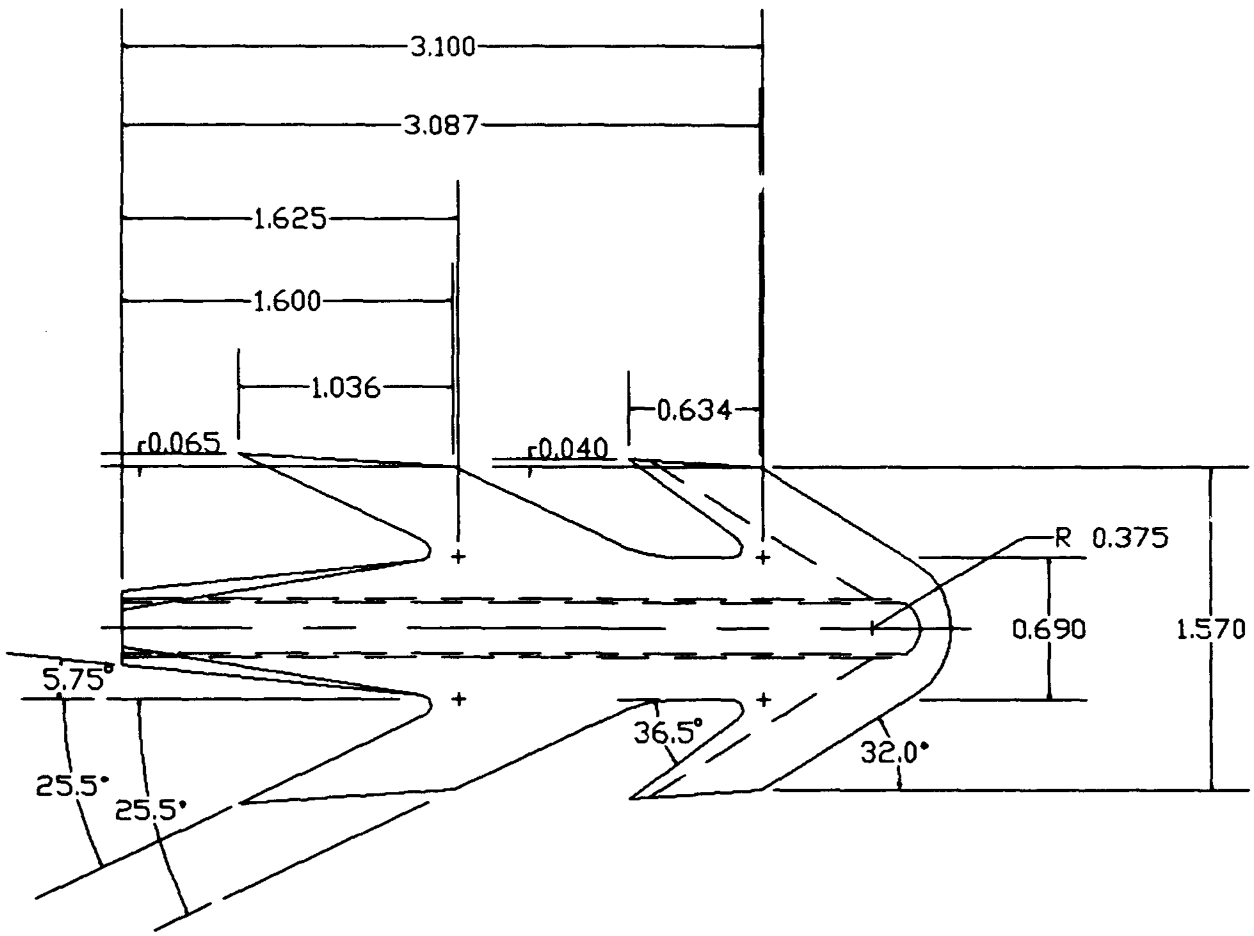

Figure 2. CCEMG armature/sabot.

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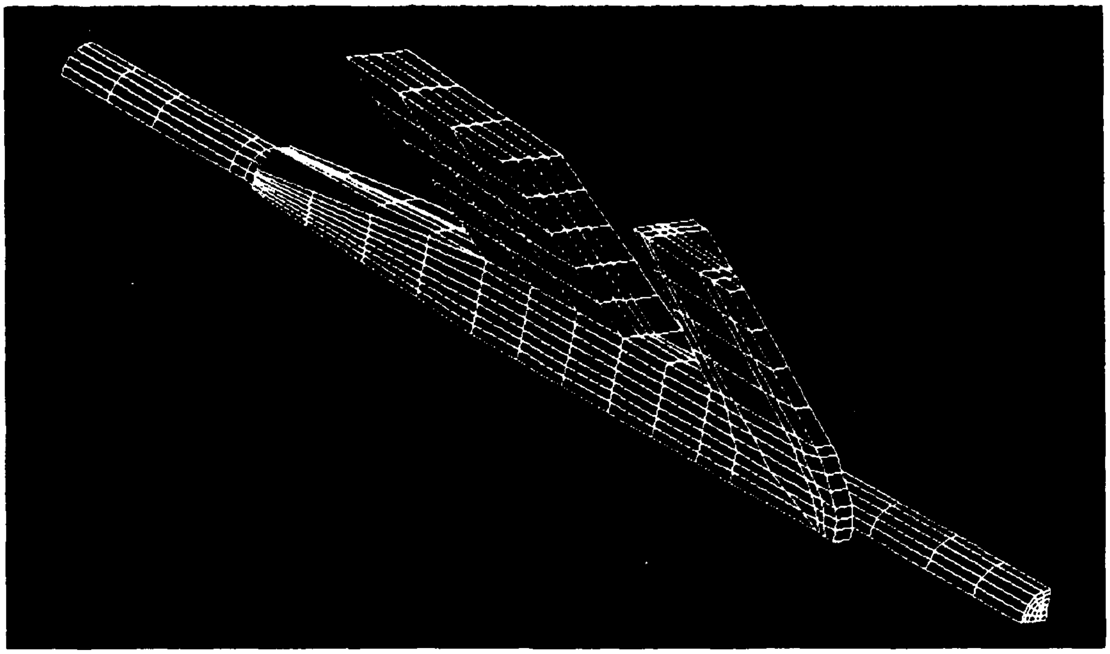

Figure 3. FE model of one-quarter of CCEMG ILP.

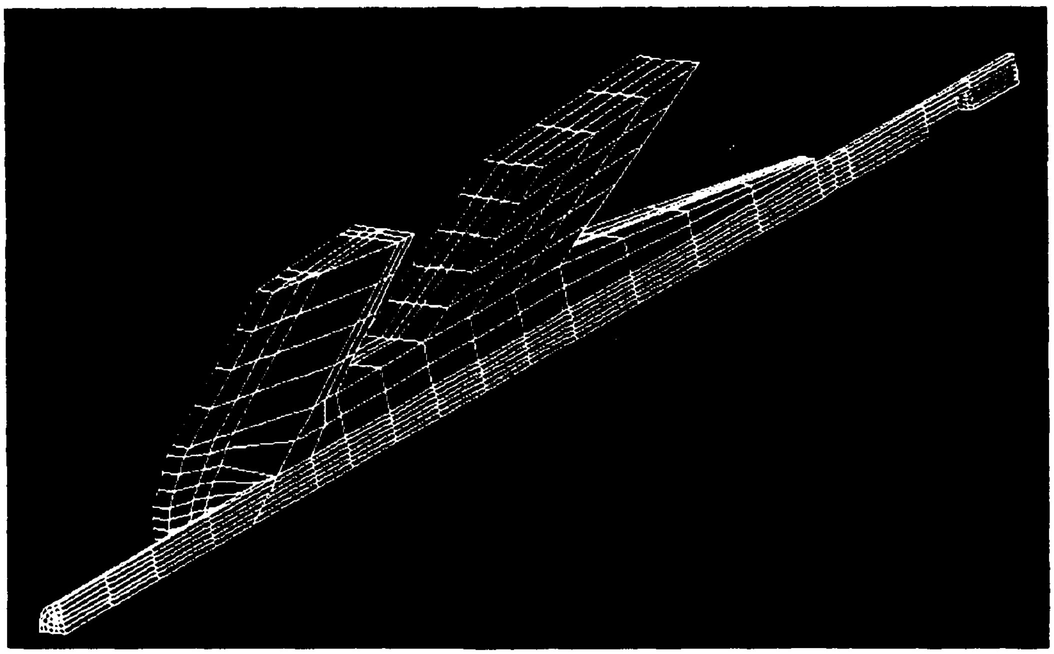

Figure 4. Reverse angle view of CCEMG ILP model.

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Table 1. Computed Mass of CCEMG ILP Components

Component	Mass (g)
Penetrator	83.75
Flare Assembly	3.85
Armature/Sabot	95.36
Total	182.96

3. ILP LOADING

The ILP was modeled using the peak loading condition which was specified by an acceleration of 190,000 g's. This inertial acceleration load was balanced against a force loading which was applied in one case via an axial directed pressure along the rear armature contact. The other load application technique used three-dimensional nodal force loads applied throughout the entire armature/sabot body.

To calculate the equivalent pressure load, Newton's second law (force equals mass times acceleration) was used. Dividing this force value by the area of the rear armature contact over which it is applied results in a pressure of 116 ksi (800 MPa) for the equivalent pressure load.

For the case of the nodal force loading, the magnitude of the total projectile load was provided by CEM-UT. Their code predicts the EM body forces for a stationary (velocity = 0) armature during the transient current pulse. Values were provided for six discrete times with the time of the peak axial force, 75.9 kips (338 kN), employed in the FE analysis. The projectile body was divided into six regions as shown in Figure 5, with each region carrying a different proportion of the overall load. The breakdown of the axial and transverse forces attributable to each region is provided in Table 2. The negative signs for the rail-to-rail and insulator-to-insulator forces denote that they are acting radially inward. Also listed in the table is the number of nodes located in each region of the FE geometry model. Lacking any further specific guidance on the force distribution, it was decided to uniformly distribute the total force across the nodes in a region. For example, region 5 has 100 nodes that carry 1/4 of the total axial force (due to quarter symmetry in the axial direction), 10.1 kips (45 kN), resulting in each node having a 101-lb (45 N) applied load. Similarly, the transverse loads are applied with 1/2 the total force per region (due to half symmetry in the transverse planes) being uniformly distributed among the nodes.

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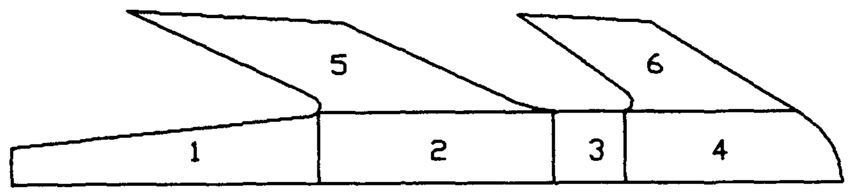

Figure 5. CCEMG ILP regions for application of body force loads.

Table 2. Body Force Distribution Through the CCEMG Armature/Sabot

	Area Number						
	1	2	3	4	5	6	Total
Axial Force (kips)	13.7	17.9	0.5	0.8	40.4	2.6	75.9
Rail-to-Rail Force (kips)	−12.5	−1.7	−0.5	−0.1	22.7	1.4	9.3
Insulator-to-Insulator Force (kips)	−1.6	−4.7	−0.1	−0.2	−12.1	−0.9	−19.6
Number of Nodes in Region	200	96	32	150	100	120	—

4. BOUNDARY CONDITIONS

When launching a projectile from an EM gun, the rails tend to be forced apart due to the in-bore EM field (Zielinski 1988). Solid armatures are generally designed with an interference with the inner rail surface in an attempt to ensure contact along the entire length of travel of the barrel (Price 1993). Previous analysis of a cannon caliber launcher had shown rail separation due to EM loading to be 0.030 in (0.762 mm) (Werst et al. 1993). However, for the bore geometry used in this analysis, an assessment of the magnitude of rail separation was not available.

Lacking a firm knowledge of what constitutes a true representation of the boundary conditions, two different conditions were applied along the top surface of the armature contacts (regions 5 and 6 in Figure 5). In the first case, the armature contacts were displaced to the nominal bore diameter. Since the armature contacts were designed with an interference fit, this boundary condition, in effect, meant that

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the leaves were compressed inward to conform with a barrel assumed to be rigid. In the second case, a zero displacement condition was assumed. This condition allowed for no radial displacement of the armature contact surfaces and was intended to replicate the condition of rail expansion with the armature contacts fully extended to maintain contact. While it is understood that neither of these conditions match actual launch conditions, their selection was intended to bound the actual launch condition that is likely to occur.

5. RESULTS

An FE analysis was performed for five cases assuming elastic material properties with various loading and boundary conditions. The first two cases examined had the equivalent pressure load of 116 ksi (800 MPa) applied to the rear armature contact. Case 1 employs the rigid barrel assumption with the top surface of the armature contacts compressed. Case 2 also has the equivalent pressure load but utilizes the zero displacement boundary condition.

Figures 6 and 7 depict the calculated stress contours for Case 1 of the penetrator and armature/sabot, respectively. Table 3 contains values of the peak through-stress of both the sabot and penetrator. Also listed is the highest localized stress value found in the sabot. While the maximum stress through the penetrator is less than its yield strength of 225 ksi (1,550 MPa), the aluminum is subjected to stresses much in excess of its 82 ksi (565 MPa) yield value. From Figure 7, it is seen that the entire rear armature is subject to failure.

Similarly, Figures 8 and 9 show the stresses for Case 2. From Figure 9, it may be noted that while the rear armature does experience through-stresses above yield, this is only true for a portion of the structure. Table 3 shows that the zero displacement boundary condition results in an alleviated stress state through the sabot.

Case 3 was run using a rigid barrel assumption with the distributed nodal force loading. Figures 10 and 11 are provided with the stress plots for the rod and sabot, respectively. Figure 11 shows that the rear armature contact has high localized stresses but no through-section stresses above yield. However, the front armature contact is subjected to a 122 ksi (838 MPa) through-stress. It was somewhat puzzling that the analysis would predict failure of the front borerider under a predominantly axial load which acts primarily, from Table 1, on the aft end. It was felt that the interference of the armature contacts with the

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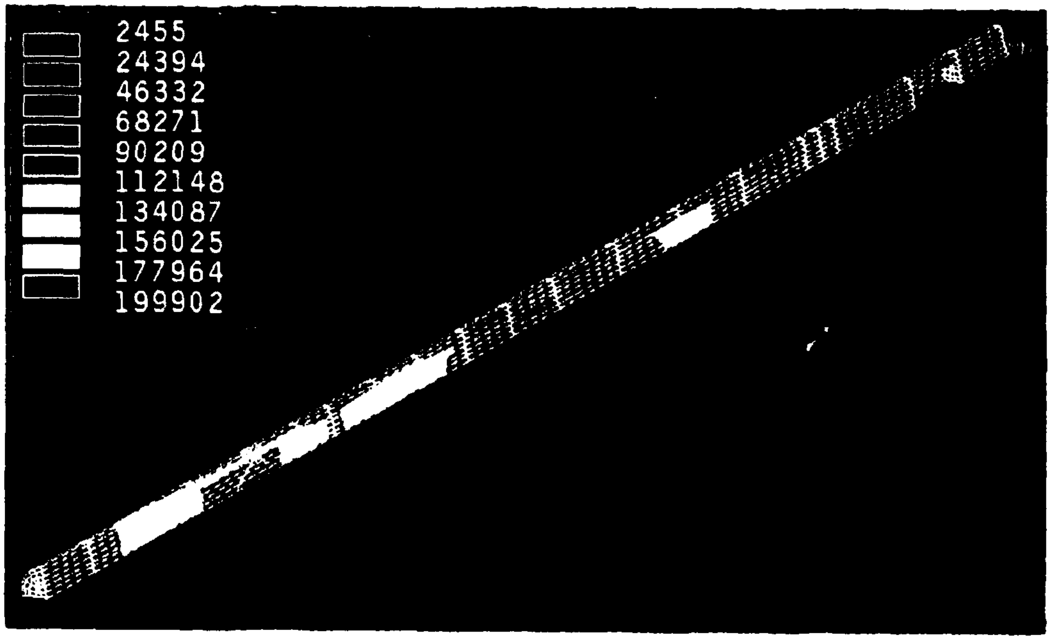

Figure 6. Stress contours in penetrator from Case 1 analysis.

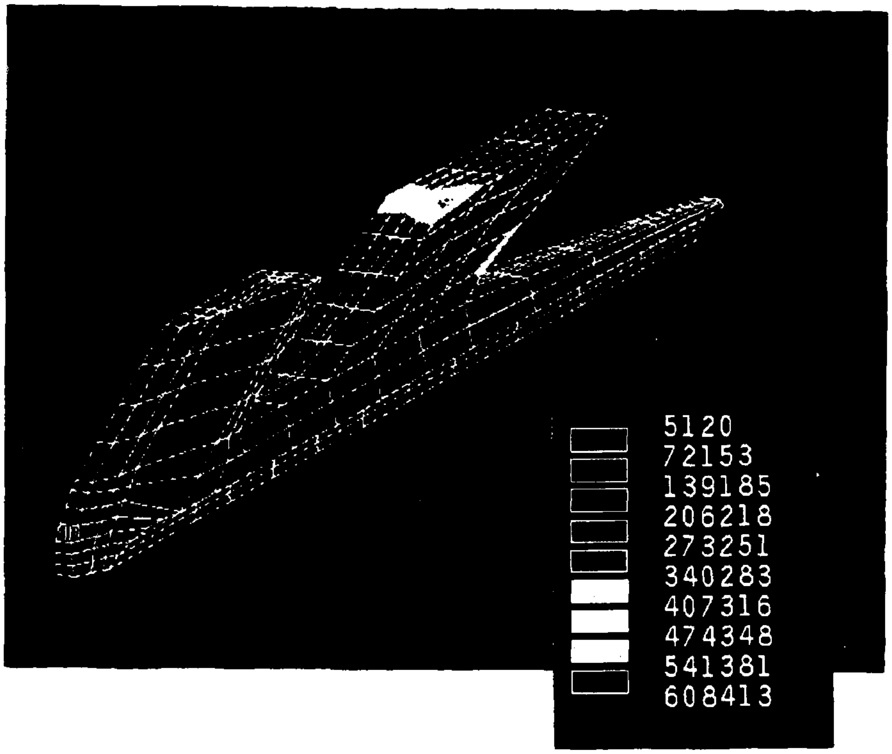

Figure 7. Stress contours in armature/sabot from Case 1 analysis.

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Table 3. Peak Stress Values Found With Various Boundary and Loading Conditions

	Loading Condition	Armature Boundary Condition	Maximum Sabot Through-Stress (ksi)	Peak Sabot Stress (ksi)	Maximum Penetrator Through-Stress (ksi)
Case 1	Pressure on Armature (P = 116,306 psi)	Armature Displaced to Bore Rail-to-Rail Dimension	273	608	200
Case 2	Pressure on Armature (P = 116,306 psi)	Zero Displacement on Top of Armature Contacts	141	251	191
Case 3	Distributed Force Throughout Sabot	Armature Displaced to Bore Rail-to-Rail Dimension	122 (Front Contact)	272	124
Case 4	No Applied Load	Armature Displaced to Bore Rail-to-Rail Dimension	91 (Front Contact)	204	73
Case 5	Distributed Force Throughout Sabot	Zero Displacement on Top of Armature Contacts	53	94	113

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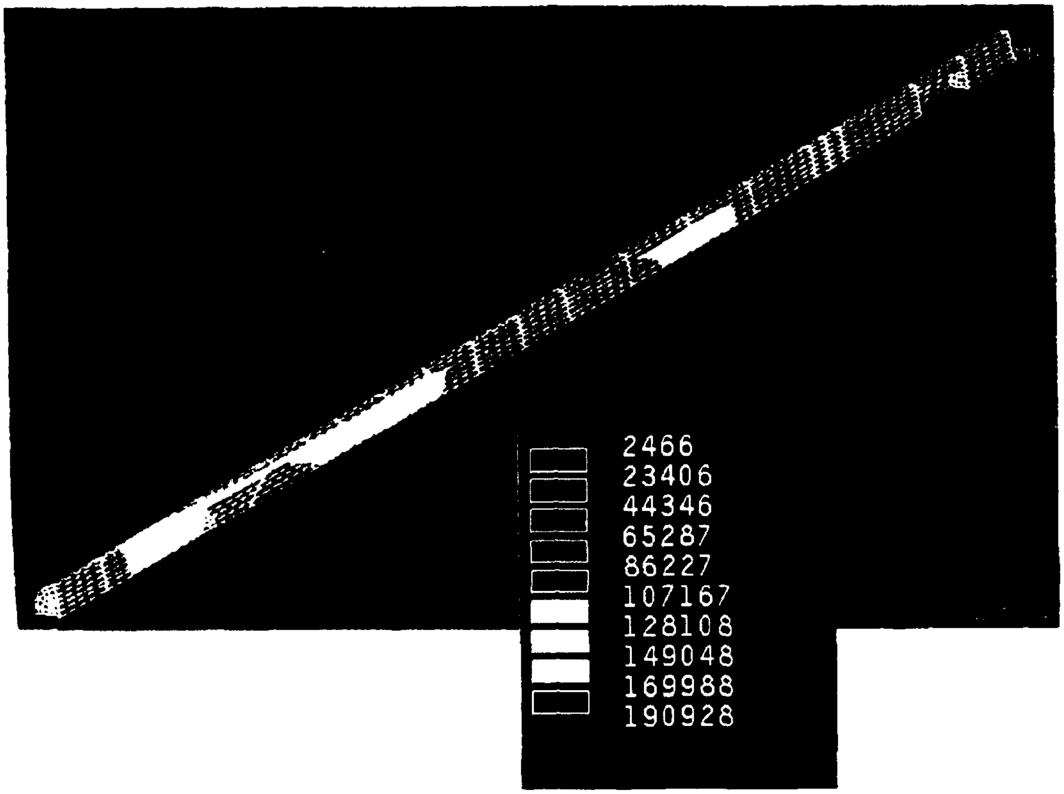

Figure 8. <u>Stress contours in penetrator from Case 2 analysis.</u>

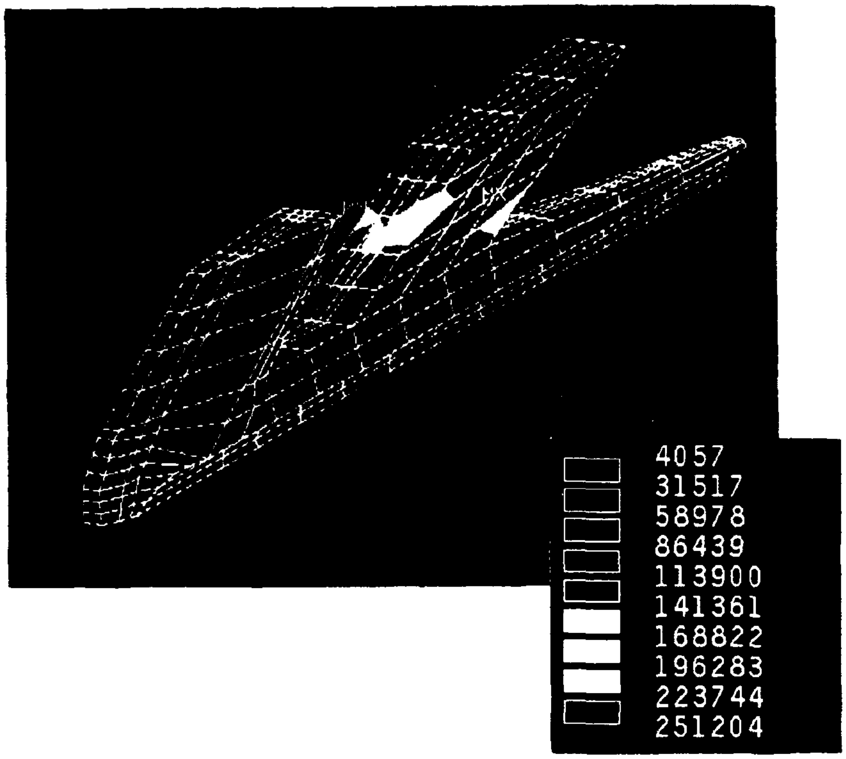

Figure 9. <u>Stress contours in armature/sabot from Case 2 analysis.</u>

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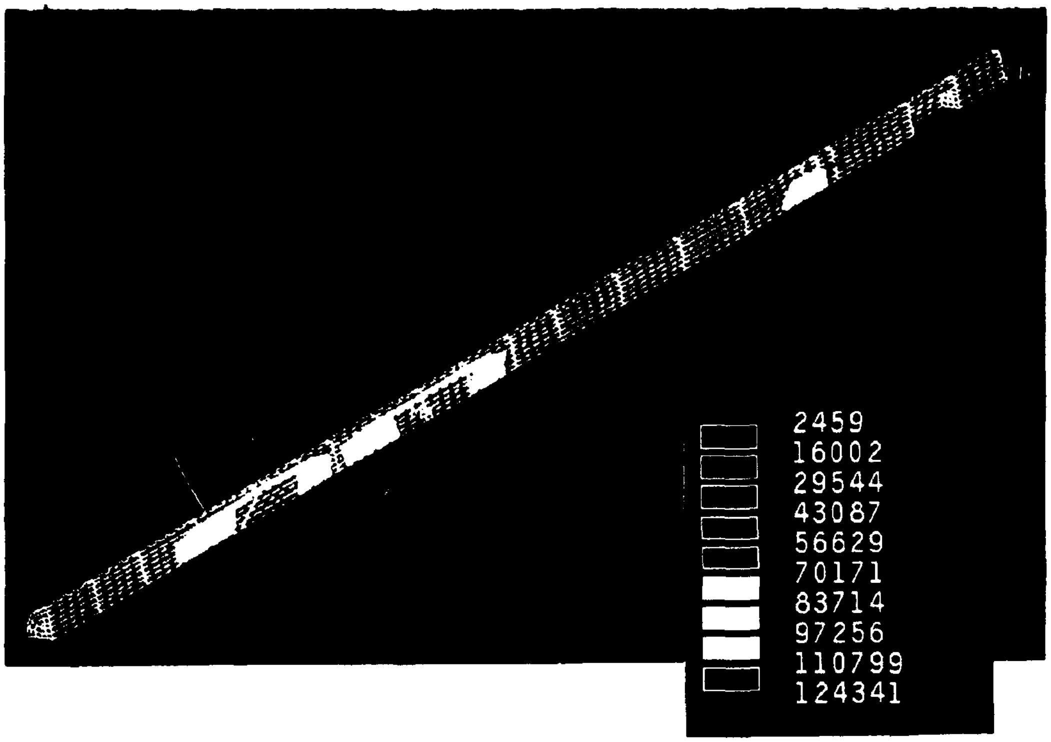

Figure 10. Stress contours in penetrator from Case 3 analysis.

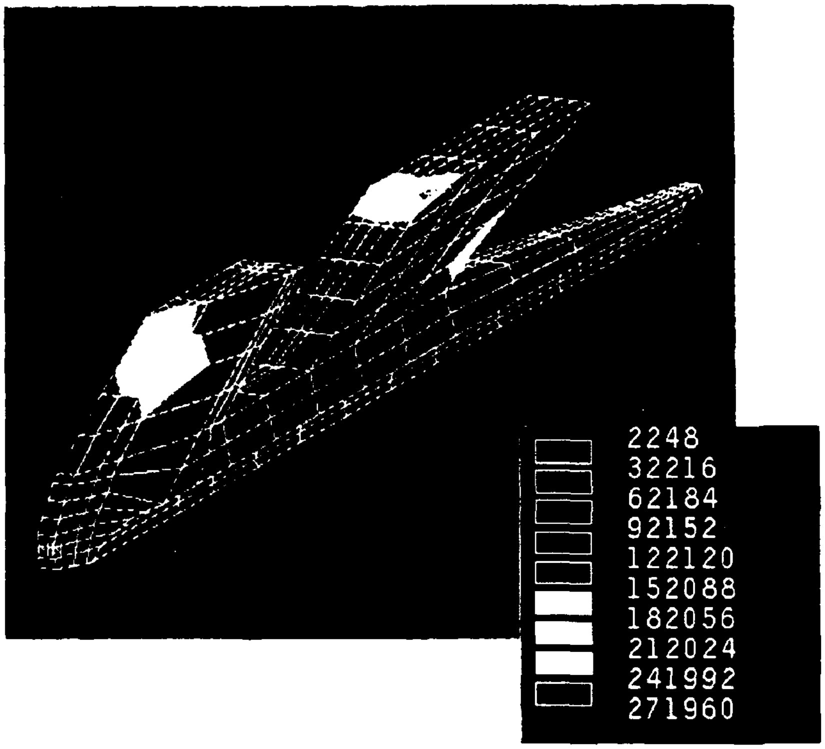

Figure 11. Stress contours in armature/sabot from Case 3 analysis.

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barrel may be excessive to the point that the armature would be incapable of withstanding the insertion force into the bore.

To check this hypothesis, Case 4 was run with no applied loading. Only the rigid barrel boundary condition was applied. Figures 12 and 13 show the stress contours for this case. From Figure 13, it is seen that high localized stresses exist in both of the armature contacts. However, only the front armature contact has a through-section stress in excess of yield, 91 ksi (627 MPa). Therefore, assuming a rigid barrel, the front armature contact would plastically deform when inserted into the bore. This result suggests that the maximum interference of the armature contacts, 0.065 in (0.165 cm) in the rear and 0.040 in (0.102 cm) in the front on the radius, is excessive and should be reduced.

Case 5 was done with the zero displacement boundary condition and the distributed nodal force loading. Figures 14 and 15 show the stress state in the projectile components, and only a small localized area at the base of the rear armature exceeds the yield value. From Table 3, it is noted that both the penetrator and sabot peak through-stresses are well below yield.

6. CONCLUSIONS

The analysis of Case 4 points out the need to reduce the interference of the armature contacts with the gun barrel. While it is conceded that the barrel does have some compliancy and will "give" some when the projectile is inserted, the fact that the results show stresses 10% over yield leads to the conclusion that the interference should be less.

Both Cases 3 and 5 resulted in very low stresses through the penetrator and rear armature contact that had acceptable stress levels. While Case 5 exhibited acceptable stress in the front armature contact, Case 3 did not. However, it is felt that the elevated front armature stresses in Case 3 are a result of the large interference with the gun bore. By reducing the interference as suggested previously, it is felt that the stress in the front armature contact would be significantly less. Therefore, based on the analysis results, it is felt that the CCEMG ILP with reduced armature contact interference is structurally robust.

The results of the analysis also point out the importance of accurately modeling the EM body force loading. Comparing Case 1 with 3 and Case 2 with 4 in Table 3, it is readily apparent that the use of an

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Figure 12. Stress contours in penetrator from Case 4 analysis.

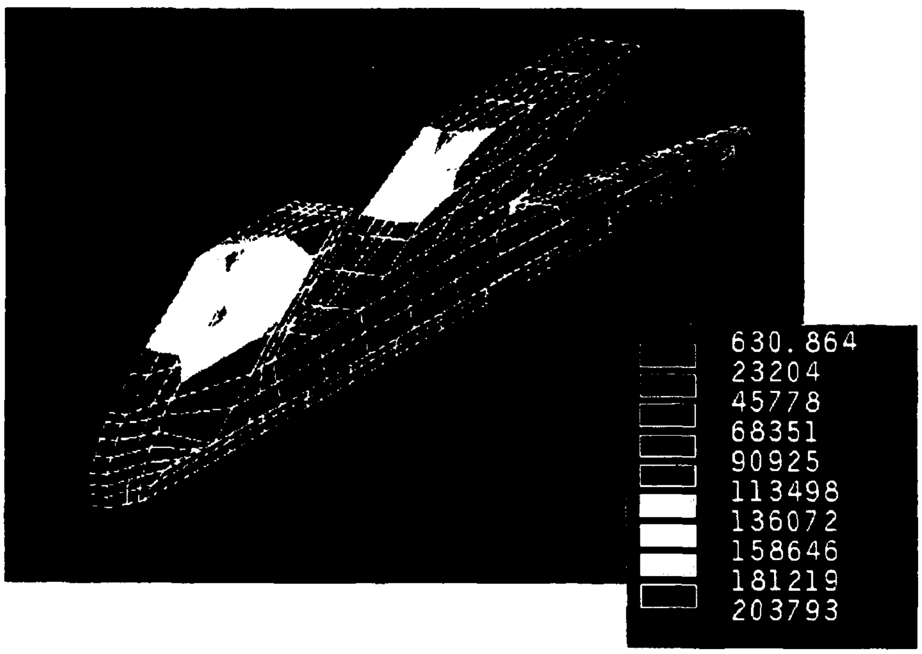

Figure 13. Stress contours in armature/sabot from Case 4 analysis.

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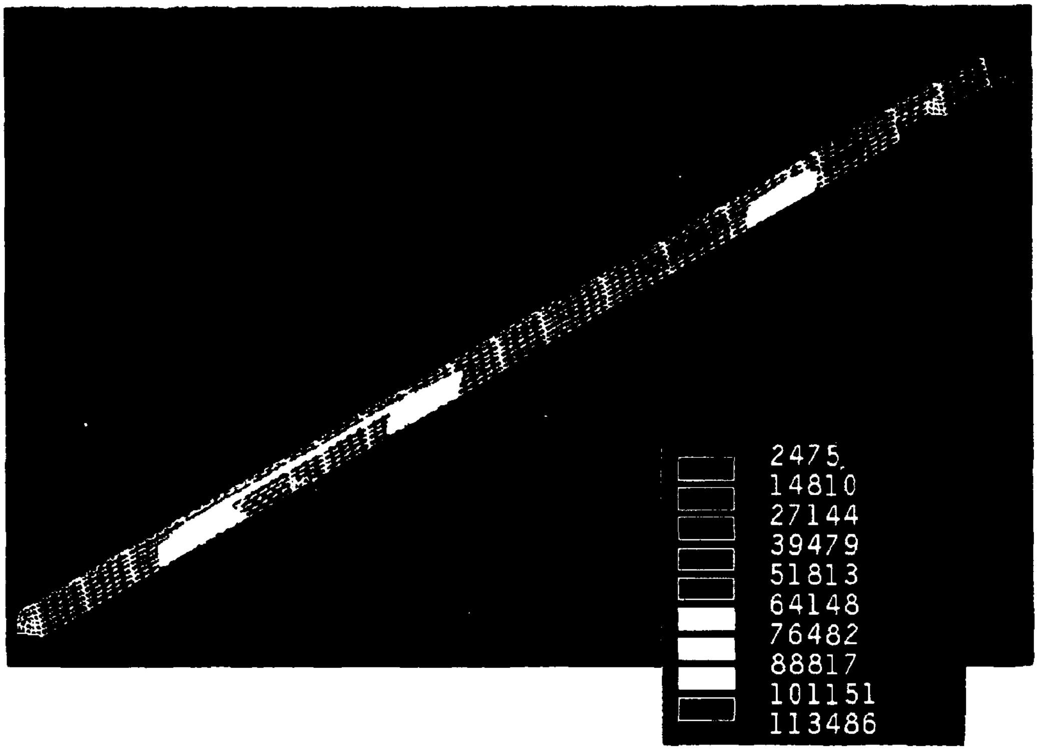

Figure 14. Stress contours in penetrator from Case 5 analysis.

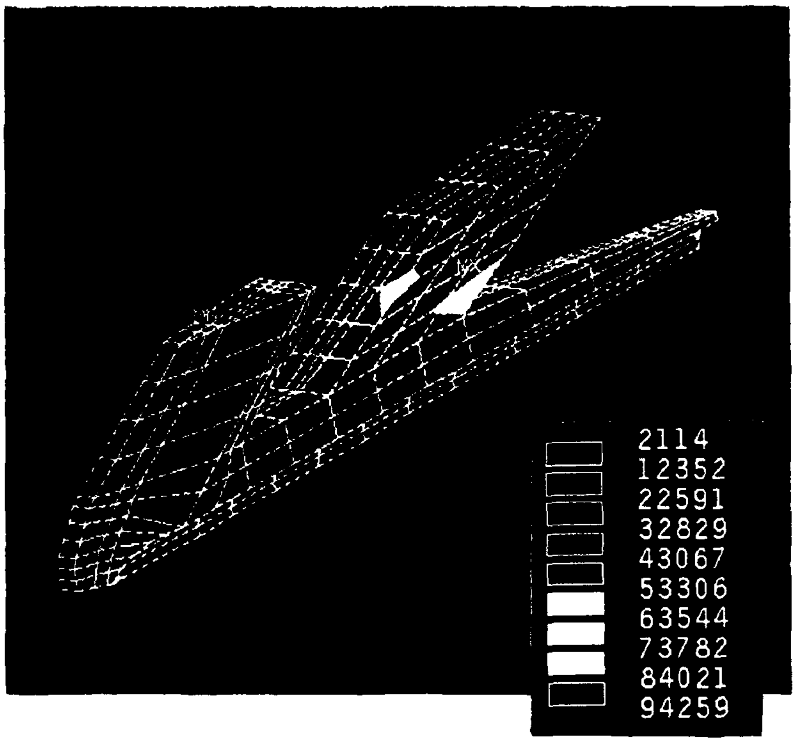

Figure 15. Stress contours in armature/sabot from Case 5 analysis.

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equivalent pressure load for a mid-riding EM armature results in significantly higher stress values. Obviously, the more precise the magnitude of the nodal force loading, the more accurate the results of the stress analysis will be. This lends credence to ongoing attempts to couple EM codes which predict current density and subsequently body forces with structural FE codes. Accomplishing this will allow for very accurate representation of the EM load on a projectile.

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